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RESEARCH TURBINE FOR HIGH-TEMPERATURE CORE ENGINE APPLICATION

II - Effect of Rotor Tip Clearance on Overall Performance

by Edward M. Szanca, Frank P. Behning, and Harold J. Schum

Lewis Research Center

Cleveland, Ohio 44135

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16. Abstract

A 25.4-cm (10-in.) tip diameter turbine was tested to determine the effect of rotor radial tip clearance on turbine overall performance. The test turbine was a half-scale model of a 50.8-cm- (20-in.-) diameter research turbine designed for high-temperature core engine application. The test turbine was fabricated with solid vanes and blades with no provision for cooling air and tested at much reduced inlet conditions. The tests were run at design speed over a range of pressure ratios for three different rotor clearances ranging from 2.3 to 6.7 percent of the annular blade passage height. The results obtained are compared to the results obtained with three other turbines of varying amounts of reaction.

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RESEARCH TURBINE FOR HIGH-TEMPERATURE CORE ENGINE APPLICATION

II - EFFECT OF ROTOR TIP CLEARANCE ON OVERALL PERFORMANCE

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SUMMARY

An experimental investigation was made of a 25.4-centimeter (10-in.) tip diameter turbine to determine the effect of increasing rotor tip clearance on overall performance. The test turbine was a half-scale model of a 50.8-centimeter- (20-in.-) diameter research turbine designed for high-temperature core engine application. The test turbine was fabricated with solid vanes and blades with no provision for cooling air and tested at much reduced inlet-air pressure and temperature.

Three radial tip clearances were investigated ranging from 2.3 to 6.7 percent of the annular blade passage height. The turbine was operated at design speed over a range of pressure ratios. Radial angle surveys at the rotor exit were taken for each clearance.

At the design radial clearance of 2.3 percent (0.0432 cm (0.017 in.)), design speed, and design specific work output, the total efficiency of the turbine was 88.1 percent. The corresponding total pressure ratio was 1.803. At this pressure ratio, as the clearance was increased, the total efficiency decreased about 1.7 percent for every 1 percent increased in radial clearance.

Comparisons of the results obtained in this investigation were compared to those of other investigations. The losses due to radial clearance for the subject turbine were greater than for the referenced impulse turbine but less than for the two reaction turbines used in the comparison.

INTRODUCTION

Future engines of the turbofan variety will have high compressor pressure ratios, high turbine inlet temperatures, and high bypass ratios. The turbines will require large amounts of coolant flow eminating from the vane, blades, stator end walls, and rotor shrouds. The core engine turbine component of these turbofans is characterized by small blade heights, high hub to tip radius ratios, low aspect ratios, and generally smaller outside diameters as compared to the turbine of pure turbojet engines designed for similar applications. As a consequence, the rotor tip clearance can be a large percentage of the blade annular passage height. The high temperatures encountered in actual engines may also make it difficult to maintain close control over radial clearances. Previous investigations (refs. 1 to 3) have shown that radial clearance losses can be appreciable and are influenced by such things as the amount of reaction across the rotor, blade hub to tip radius ratio, and blade aspect ratio. Also, it is reasonable to assume that the coolant flow from the rotor shroud may effect the radial clearance losses. Therefore, it was desirable to determine the penalty associated with varying the radial clearances that may be encountered in the high-temperature application.

Prior to testing the turbine at the high inlet conditions encountered in actual engine operation, a full-scale 50. 8-centimeter- (20-in. -) diameter turbine was designed and built under contract to determine the effects of large amounts of coolant flow on aerodynamic performance. This turbine was designed to operate at an inlet temperature of 2200 K (3960° R) and will be tested at a reduced inlet temperature of 783 K (1410° R). However, the amount of coolant flow and the temperature ratio of primary air to coolant air for the reduced temperature test will simulate that of the high-temperature turbine. To determine the base performance of the cooled turbine, a half-scale turbine was designed and fabricated with solid vanes and blades of the same aerodynamic profile but with no provisions for cooling air. This turbine was tested over a range of speed and pressure ratios and the results reported in reference 4. This turbine also provides the basic configuration for the radial clearance study which is the subject of this report.

The subject turbine has a tip diameter of 25. 4 centimeters (10 in.), a rotor blade height of 1. 905 centimeters (0. 75 in.), and a rotor tip clearance of 0. 0432 centimeter (0. 017 in.). This clearance represents 2. 3 percent of the blade annular passage height. Two additional clearance values were investigated by successively removing metal from the tips of the rotor blades. The additional clearance values were 0. 0635 and 0. 1270 centimeter (0. 025 and 0. 050 in.) which represented 3. 3 and 6. 7 percent of the blade annular height, respectively.

All tests were conducted at an inlet pressure of 17. 237 newtons per square centimeter (25.0 psia), an inlet temperature of approximately 306 K (550° R), and design equivalent speed. For each clearance the turbine was tested over a range of total pressure ratio from 1. 3 to 2. 8. Rotor exit angle radial survey were also taken for each clearance value. The effect of clearance on performance is presented in terms of weight flow, specific work output, exit flow angle, and turbine efficiency. In addition, the results of clearance as a function of efficiency are compared to the experimental results of references 1 to 3.

SYMBOLS

b vane or blade height, cm; in.

c chord length, cm; in.

D diameter, cm; in.

Δh specific work, J/g; Btu/lb

N turbine shaft speed, rpm

p pressure, N/cm^2 abs; psia

r radius, cm; in.

s pitch, cm; in.

T absolute temperature, K; ^{o}R

U blade velocity, m/sec; ft/sec

V absolute gas velocity, m/sec; ft/sec

W relative gas velocity, m/sec; ft/sec

w mass flow, kg/sec; lb/sec

α absolute gas flow angle measured from the axial direction, deg

γ ratio of specific heat

δ ratio of inlet total pressure to U.S. standard sea-level pressure, p_0/p^*

ϵ function of γ used in relating parameters to those using air inlet conditions at U.S. standard sea-level conditions, $\frac{\gamma^*}{\gamma}\left[\left(\frac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)} \Bigg/ \left(\frac{\gamma^*+1}{2}\right)^{\gamma^*/(\gamma^*-1)}\right]$

η_s static efficiency (based on inlet total- to exit-static pressure ratio)

η_t total efficiency (based on inlet total- to exit-total pressure ratio)

θ_{cr} squared ratio of critical velocity at turbine inlet to critical velocity at U.S. standard sea-level air, $\left(V_{cr}/V^*_{cr}\right)^2$

τ torque, N-m; ft-lb

ω turbine speed, rad/sec

Subscripts:

cr condition corresponding to Mach number of unity

b blade

h hub section

m mean section

t tip section

v vane

x axial component

0 station at turbine inlet (fig. 8)

1 station at stator exit (fig. 8)

2 station at turbine exit (fig. 8)

Superscripts:

' absolute total state

* U. S. standard sea-level conditions (temperature equal to 288.15 K (518.7° R), pressure equal to 10.13 N/cm^2 abs (14.7 psia))

TURBINE DESCRIPTION

The selected design criteria, which are typical for the first stage of a two-stage high-temperature axial flow core engine turbine are summarized in table I. The hot engine conditions are for a full-scale turbine using an ASTM-A-1 fuel to air ratio of 0.0435 while the equivalent conditions are given for a half-scale model of this turbine. The numbers in the table do not include the coolant flow rates nor the effect of coolant flow on turbine performance.

The velocity diagram evolved to meet the design aerodynamic requirements is shown in figure 1. All quantities represent the free stream uniform flow conditions. Pertinent test-turbine geometry are presented in table II.

The test-turbine stator was fabricated with untwisted vanes of constant mean-section profile, ignoring the relatively small amount of twist (about 3°) from hub to tip if designed for free-vortex conditions. The rotor blades were designed for free-vortex flow and simple radial equilibrium. The subject investigation was conducted using solid (uncooled) vanes and blades. The vanes and blades for the turbine are characterized by blunt leading and trailing edges, low aspect ratio, and high thickness to chord ratios. All of these geometric factors are considered detrimental to high turbine efficiency.

The vane and blade design surface pressure distributions at the mean section are taken from reference 4 and shown in figures 2 and 3, respectively.

Vane profile coordinates are presented in table III; rotor blade coordinates at the hub, mean, and tip section are given in table IV. All dimensions are for the half-scale test model. A photograph of the stator is presented in figure 4. Similarly, a photograph of the rotor assembly is shown in figure 5.

APPARATUS, INSTRUMENTATION, AND PROCEDURE

Apparatus

The apparatus consisted of the turbine as described in the preceding section, a cradled dynamometer to absorb the power output of the turbine while controlling its speed and an inlet and exhaust piping system with flow controls. The arrangement of the experimental equipment is shown schematically in figure 6. The turbine was driven by dry pressurized air supplied from the laboratory air system. Before reaching the turbine, the air was filtered, and measured with a calibrated flat-plate orifice. Metering of the air into the turbine was accomplished by a remotely controlled pressure regulating valve. After flowing through the turbine, the air was exhausted into the laboratory altitude exhaust system. Turbine exhaust pressure was controlled by a remotely operated butterfly valve in the exhaust piping.

A 224-kilowatt (300-hp) cradled dynamometer was used to absorb turbine shaft output power, control speed, and measure torque. The dynamometer was coupled to the turbine shaft through a gearbox, which provided relative rotative speeds between dynamometer and turbine of 1.0 to 4.25. The dynamometer and gearbox frames were bolted together and the entire assembly floated on hydrostatic oil bearings. Figure 7 is a photograph of the test facility.

Instrumentation

A schematic cross section of the turbine is shown in figure 8. Also shown on this figure are the instrument measuring stations.

At the turbine inlet (station 0), the instrumentation consisted of static pressure and total temperature measuring devices. The temperature was measured with three thermocouple rakes, each containing two thermocouples located at the area center radii of two equal annular areas. Static pressure was obtained from eight taps with four on the inner wall and four on the outer wall. The inner and outer taps were located opposite each other and were spaced 90° apart about the circumference.

The instrumentation at station 1, between the stator and the rotor, consisted of eight static pressure taps. These taps were spaced 90° apart with four on the inner and four on the outer as described for station 0.

At the turbine outlet, station 2, measurements of static pressure, total pressure, total temperature, and flow angle were made. The static pressure was measured with eight wall taps located as described for stations 0 and 1. The outlet flow angle was measured with an angle sensitive probe and a self-alining probe actuator. This probe was also used to obtain the measured flow angle from the rotor hub to the rotor tip at the turbine exit. Also mounted on the angle measuring probe were provisions for determining total pressure and total temperature.

Turbine torque was transmitted to a commercial strain-gage load cell through a torque arm attached to the dynamometer stator.

The rotational speed was detected by a magnetic pickup and shaft mounted gear.

All pressures were measured with calibrated electrical transducers. A 200 channel data acquisition system was used to measure and record the electrical signals from the transducers.

Procedure

Turbine performance was obtained at nominal turbine inlet conditions of 17.237 newtons per square centimeter (25.0 psia) pressure and ambient temperature (approximately 306 K or 550° R). Data were obtained at design equivalent speed and over a range of inlet- to exit-total pressure ratios from 1.3 to 2.8. A rotor exit survey of flow angle was made at design equivalent speed and at a number of pressure ratios bracketing the design pressure ratio. The variation in angle with radius ratio at design equivalent speed and specific work output was determined from cross plots of the aforementioned data.

Tests conducted with this turbine at the design rotor blade radial clearance of 2.3 percent (0.0432 cm or 0.017 in.) have been reported in reference 4. The results of these tests at design speed are repeated in this report. Two other clearances were investigated, these were 3.3 percent (0.0635 cm or 0.025 in.) and 6.7 percent (0.1270 cm or 0.050 in.). The clearances were increased after each series of tests by machining the necessary material from the rotor tips until the desired clearance was obtained.

Torque calibrations were obtained before and after each performance run. The calibrations were obtained with the dynamometer in the motoring mode, rotating the turbine at a speed of approximately 4500 rpm. Thus, most of the effects of bearing and seal friction were accounted for in the calibration. Turbine windage losses were minimized during the calibrations by evacuating the air from the turbine housing.

The turbine was rated on the basis of inlet- to exit-total pressure ratio efficiency. Inlet and outlet total pressures were calculated from mass flow, static pressure, total temperature, and flow angle. At the inlet, the flow angle α was assumed to be axial. At the exit, the total temperature was computed from measured values of inlet total temperature and the specific work output of the turbine.

RESULTS AND DISCUSSION

The results of this investigation are presented in three sections. The first section describes the overall performance of the turbine with the design radial clearance of 2.3 percent (0.0432 cm or 0.017 in.) over a range of speeds and pressure ratios. The second section compares turbine performance at design speed for three radial clearances: 2.3 percent (0.0432 cm or 0.017 in.), 3.3 percent (0.0635 cm or 0.025 in.), and 6.7 percent (0.1270 cm or 0.050 in.). Results of radial surveys of flow angle at the rotor exit are also presented. The third section is a comparison of efficiency loss as a function of radial clearances for the subject turbine as compared to some results of previous studies.

All results are presented in terms of air equivalent values.

Turbine Overall Performance

The overall performance map, torque curves, and mass flow curve for the subject turbine (figs. 9 to 11) were originally reported in reference 4 and are repeated here for completeness. The reference total pressure ratio, however, was changed from 1.817 as reported in reference 4 to 1.803 in the subject report. This latter pressure ratio corresponds to the point at which the design equivalent specific work of 39.572 joules per gram (17.00 Btu/lb) is obtained with the design radial clearance 0.432 centimeter (0.017 in.). The total efficiency at this pressure ratio is 88.1 percent as compared to 88.0 percent reported in reference 4. The mass flow and torque at design speed for a pressure ratio of 1.803 are 1.187 kilogram per second (2.617 lb/sec) and 36.268 newton-meter (26.75 ft-lb), respectively.

Turbine Performance with Increased Rotor Tip Clearance

<u>Mass flow characteristics</u>. - The mass flow $\epsilon w\sqrt{\theta_{cr}}/\delta$ characteristics of the subject turbine for three values of radial clearance are shown in figure 12. All three curves are plotted as a function of overall total pressure ratio and at design equivalent speed. The mass flow curve for the turbine with the 0.0432-centimeter (0.017-in.) rotor tip radial clearance is taken from figure 11.

At any given pressure ratio, the mass flow increased with radial clearance. This was due to the increased through flow over the rotor tip as a result of increased clearance. A comparison of the curves also shows that as the radial clearance increased, choking occurred at a higher overall pressure ratio.

At a total pressure ratio of 1.803 the mass flows for the various clearances are summarized in table V.

The data from table V was plotted in figure 13 and shows the change in equivalent mass flow (expressed as a percentage of the mass flow obtained at the lowest tip clearance value) plotted against tip clearance as a percentage of the blade annular passage height. This figure shows that the mass flow increased by about 0.95 percent as the radial clearance was increased from 0.0432 to 0.1270 cm (0.017 to 0.050 in.). This represents an increase in mass flow of 0.22 percent for an increase in tip clearance of 1 percent of blade annular passage height.

Specific work characteristics. - The variation of equivalent specific work output, $\Delta h'/\theta_{cr}$, with overall pressure ratio, p_0'/p_2', for three values of radial clearance are shown in figure 14. As in the case of the mass flow curves, the data is for design speed only.

Figure 14 shows that at any given pressure ratio, the specific work output decreased as the radial clearance increased. The reduction in specific work output with radial clearance increases will be discussed in more detail in the next section on rotor exit angle surveys.

At a total pressure ratio of 1.803, the equivalent specific work output for the three radial clearances are summarized in table VI.

The data from table VI was plotted in figure 15 and shows the change in specific work output (expressed as a percentage of the specific work output at the lowest tip clearance as a percentage of the blade annular passage height. This figure shows that the specific work output decreased by 7.4 percent as the radial clearance was increased from 0.0432 to 0.1270 cm (0.017 to 0.050 in.). This represents a decrease in specific work output of 1.7 percent for an increase in tip clearance of 1 percent of blade annular passage height.

Rotor exit radial surveys. - Figure 16 shows the results of a radial survey of turbine exit flow angle at design equivalent speed and a pressure ratio of 1.803 for three radial clearances. These angle surveys were conducted at one common circumferential position and are plotted against radius ratio.

The results of the survey show that the rotor exit angle near the tip changes significantly as the radial clearance is increased. At the smallest rotor tip clearance of 2.3 percent, the maximum measured exit flow angle was 5.0° near the tip. At the largest tip clearance of 6.7 percent the maximum measured exit flow angle was about 24.0° from the axial direction. The increases in angle persisted from the tip section to about the mean section for the two smaller clearances. The effect of the largest clearance was felt along the entire blade span although much more significantly near the tip.

The increased positive angle at the rotor exit as radial clearance was increased is an indication of less turning of the mass flow near the rotor tip. This underturning was due to greater unguided through flow over the rotor tip and to greater tip leakage flow over the blade tip from suction to pressure surface. The net result was an increase in the exit swirl velocity toward the positive direction which reduces the work output of the turbine and increases tip clearance losses. This was evidenced by a reduction in the torque as the radial clearance was increased as mentioned earlier and shown in figure 14.

Turbine efficiency comparison. - At design speed and a pressure ratio of 1.803 the total efficiencies obtained for the three radial clearances are summarized in table VII. The degradation in efficiency as radial clearance was increased is a direct result of increase in mass flow, the decrease in torque due to the underturning near the blade tip region and higher losses.

The data from table VII was plotted in figure 17 and shows the change in total efficiency (expressed as a percentage of the efficiency obtained at the lowest tip clearance value) plotted against tip clearance as a percentage of the blade annular passage height. This figure shows that the efficiency decreased by 7.3 percent as the radial clearance was increased from 0.0432 to 0.1270 centimeter (0.017 to 0.050 in.). This represents a decrease of about 1.7 percent for an increase in tip clearance of 1 percent of blade annular passage height.

Static pressure comparison. - The static pressure variation through the turbine at design speed and an overall total pressure ratio of 1.803 for the three clearances investigated is shown in figure 18. The pressure is presented as the ratio of the local static pressure at either the hub or tip over the total pressure at the turbine inlet.

For all clearances, the static pressures at the inlet to the turbine as well as the exit of the turbine remained relatively constant. Between the stator and rotor however, an increase in clearance resulted in a static pressure drop at both the hub and tip position of the blading.

The increase in pressure ratio across the stator as the radial clearance was increased was the result of decreased rotor tip reaction due to the increase in unguided flow area over the blade tip as the tip clearance was increased.

Comparison of Results with Other Turbine Investigation

A comparison of efficiency loss with radial clearance for various turbines is shown in figure 19. The comparisons were made on the basis of static efficiencies since this was the only information available in the references. The efficiencies are expressed as the ratio of efficiency degradation over efficiency at the minimum or design clearance. The clearance is expressed as a percent of annular passage height.

Included on the curve are a single-stage impulse turbine (ref. 1) and two turbines with rotor blade row reaction (refs. 2 and 3). In this report, reaction is defined as $1 - (W_1/W_2)^2$.

As expected, the turbines with the highest reaction showed the greatest loss in efficiency with increasing radial clearance. For example, the turbine of reference 2, which was designed for a blade row reaction of 0.862 at the tip, showed a loss of 3.7 points in efficiency for every 1 percent of radial clearance. The turbine of reference 3 which was designed for a blade row reaction of 0.834 at the tip section showed a 3.2 point drop in efficiency for each 1 percent increase in radial clearance. By contrast, the impulse turbine of reference 1 experienced a loss of 1.75 points in efficiency for every 1 percent increase in radial clearance. The subject turbine, which was designed for a reaction of 0.805 at the tip section, showed a loss in efficiency 1.9 points for every every 1 percent increase in radial clearance.

It would be expected that the effect of tip clearance on mass flow would be governed by the degree of rotor tip reaction being greater for higher reaction.

SUMMARY OF RESULTS

An experimental investigation of a 25.4-centimeter (10-in.) tip diameter turbine was made to determine the effect of rotor radial tip clearance on turbine overall performance. The test turbine was a half-scale model of a 50.8-centimeter (20-in.) research turbine designed for high-temperature core engine application. The test turbine was fabricated with solid vanes and blades with no provision for cooling air.

Three radial clearances were investigated. These were 2.3, 3.3, and 6.7 percent of the annular blade passage height. For each clearance, data were taken at design speed over a range of inlet- to exit-total pressure ratios of 1.3 to 2.8. All performance parameters are presented in equivalent terms based upon standard sea-level conditions.

At the design radial clearance of 2.3 percent (0.0432 cm or 0.017 in.), design speed, and an overall pressure ratio of 1.803, the total efficiency of the subject turbine was 88.1 percent. At this pressure ratio the corresponding specific work output was 39.572 joules per gram (17.00 Btu/lb). All comparisons of performance among the three radial clearances were at design speed, a pressure ratio of 1.803, and are as follows:

1. The total efficiency and specific work output decreased linearly with increasing radial tip clearance. For every 1 percent increase in radial clearance, both decreased about 1.7 percent.

2. The mass flow increased linearly with increasing radial tip clearance. For every 1 percent increase in radial clearance the mass flow increased 0.22 percent.

3. Radial surveys taken at the turbine exit showed that the exit flow angle varied as the rotor tip clearance was increased. The most noticeable changes occurred between the mean section and tip section of the blades where large underturning was observed.

4. Comparisons of the results of this investigation with those of three reference turbines indicate similar effects of clearance on turbine efficiency. The turbines with reaction through the rotor blade experienced a greater loss in efficiency with increasing radial clearance than did the impulse turbine. The subject turbine, whieh was designed for a moderate amount of reaction showed a loss in efficiency somewhat greater than the impulse turbine but less than that for the two referenced reaction turbines.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 30, 1973,
501-24.

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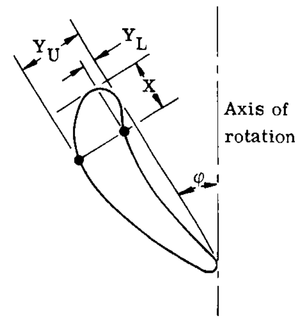
TABLE I. - TURBINE DESIGN OPERATING VALUES

Performance parameter	Hot engine conditions (ASTM-A-1/Air = 0.0435)	Air-equivalent conditions
Tip diameter, D_t, cm (in.)	50.8 (20.0)	25.4 (10.0)
Inlet total temperature, T_0', K (^oR)	2200 (3960)	288.2 (518.7)
Inlet total pressure, p_0', N/cm^2 abs (psia)	386.1 (560.0)	10.13 (14.7)
Mass flow, w, kg/sec (lbm/sec)	63.82 (140.72)	1.207 (2.660)
Turbine rotative speed, N, rpm	16 687	12 388
Specific work output, $\Delta h'$, J/g (Btu/lbm)	287.25 (123.4)	39.572 (17.00)
Mean blade speed, U_m, m/sec (ft/sec)	410.6 (1347)	152.4 (500.0)
Inlet- to exit-total pressure ratio, p_0'/p_2'	-------------	1.818
Total efficiency, η, percent	-------------	0.87

TABLE II. - TEST TURBINE GEOMETRY

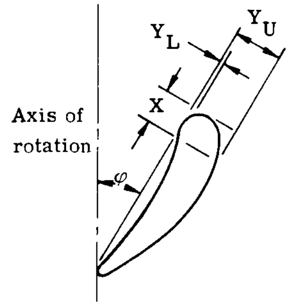
Stator	
Mean diameter, D_{mv}, cm (in.)	23.495 (9.25)
Vane height, b_v, cm (in.)	1.905 (0.75)
Axial chord, c_{xv}, cm (in.)	1.905 (0.75)
Axial solidity, $(c_{xv}/s_v)m$	0.929
Aspect ratio, b_v/c_x	1.000
Number of vanes	36
Leading edge radius, cm (in.)	0.254 (0.100)
Trailing edge radius, cm (in.)	0.0445 (0.0175)
Rotor	
Mean diameter, D_{mb}, cm (in.)	23.495 (9.25)
Blade height, b_b, cm (in.)	1.905 (0.75)
Axial chord, c_{xb}, cm (in.)	1.715 (0.675)
Axial solidity, $(c_{xb}/s_b)m$	1.652
Aspect ratio, b_b/c_{xb}	1.111
Number of blades	64
Tip clearance, cm (in.)	0.043 (0.017)
Leading edge radius, cm (in.)	0.1494 (0.0588)
Trailing edge radius, cm (in.)	0.0445 (0.0175)

TABLE III. - STATOR VANE COORDINATES



Mean Section					
Orientation angle, φ, deg					
44°37'					
X		Y_L		Y_U	
cm	in.	cm	in.	cm	in.
0	0	0.2540	0.100	0.2540	0.100
.064	.025	------	------	.4255	.1675
.127	.050	------	------	.5029	.1980
.191	.075	------	------	.5613	.2210
.254	.100	------	------	.6071	.2390
.318	.125	------	------	.6439	.2535
.381	.150	.0318	.0125	.6731	.2650
.445	.175	.0597	.0235	.6960	.2740
.508	.200	.0826	.0325	.7137	.2810
.572	.225	.1029	.0405	.7246	.2860
.635	.250	.1207	.0475	.7341	.2890
.699	.275	.1334	.0525	.7366	.2900
.762	.300	.1461	.0575	.7379	.2905
.889	.350	.1651	.0650	.7315	.2880
1.016	.400	.1803	.0710	.7163	.2820
1.143	.450	.1880	.0740	.6909	.2720
1.270	.500	.1918	.0755	.6617	.2605
1.397	.550	.1905	.0750	.6287	.2475
1.524	.600	.1867	.0735	.5944	.2340
1.651	.650	.1765	.0695	.5550	.2185
1.778	.700	.1651	.0650	.5156	.2030
1.905	.750	.1486	.0585	.4724	.1860
2.032	.800	.1308	.0515	.4255	.1675
2.159	.850	.1105	.0435	.3734	.1470
2.286	.900	.0902	.0355	.3188	.1255
2.413	.950	.0660	.0260	.2591	.1020
2.540	1.000	.0406	.0160	.1956	.0770
2.667	1.050	.0127	.0050	.1295	.0510
2.776	1.093	.0445	.0175	.0445	.0175
		Stacking axis coordinates			
		x		y	
		cm	in.	cm	in.
		2.7318	1.0755	0.0445	0.0175

TABLE IV. - ROTOR BLADE COORDINATES



X		Hub section				Mean section				Tip section			
		Orientation angle, φ, deg											
		$17^{\circ}54'$				24°				$30^{\circ}13'$			
		Y_L		Y_U		Y_L		Y_U		Y_L		Y_U	
cm	in.	cm	in.	cm	in.	cm	in.	cm	in.	cm	in.	cm	in.
0	0	0.1494	0.0588	0.1494	0.0588	0.1494	0.0588	0.1494	0.0588	0.1494	0.0588	0.1494	0.0588
.064	.025	------	------	.3239	.1275	------	------	.3302	.1300	------	------	.3023	.1190
.127	.050	------	------	.4318	.1700	------	------	.4229	.1665	------	------	.3874	.1525
.191	.075	------	------	.5080	.2000	------	------	.4877	.1920	------	------	.4483	.1765
.254	.100	------	------	.5626	.2215	.0419	.0165	.5448	.2145	.0419	.0165	.4978	.1960
.381	.150	.1422	.0560	.6553	.2580	.1321	.0520	.6223	.2450	.1257	.0495	.5664	.2330
.508	.200	.2210	.0870	.7125	.2805	.2007	.0790	.6655	.2620	.1803	.0710	.6007	.2365
.635	.250	.2794	.1100	.7417	.2920	.2515	.0990	.6795	.2675	.2184	.0860	.6071	.2390
.762	.300	.3137	.1235	.7442	.2930	.2807	.1105	.6693	.2635	.2426	.0955	.5931	.2335
.889	.350	.3353	.1320	.7201	.2835	.2921	.1150	.6414	.2525	.2464	.0970	.5664	.2230
1.016	.400	.3353	.1320	.6782	.2670	.2896	.1140	.6007	.2365	.2388	.0940	.5283	.2080
1.143	.450	.3175	.1250	.6236	.2455	.2718	.1070	.5486	.2160	.2210	.0870	.4801	.1890
1.270	.500	.2870	.1130	.5601	.2205	.2388	.0940	.4877	.1920	.1943	.0765	.4305	.1695
1.397	.550	.2413	.0950	.4890	.1925	.1956	.0770	.4191	.1650	.1575	.0620	.3747	.1475
1.524	.600	.1854	.0730	.4026	.1585	.1461	.0575	.3454	.1360	.1168	.0460	.3099	.1270
1.651	.650	.1219	.0480	.3073	.1210	.0927	.0365	.2629	.1035	.0737	.0290	.2375	.0935
1.778	.700	.0533	.0210	.2032	.0800	.0356	.0140	.1715	.0675	.0279	.0110	.1575	.0620
1.905	.750	------	------	------	------	.0445	.0175	.0445	.0175	------	------	------	------
1.908	.751	------	------	------	------	------	------	------	------	.0445	.0175	.0445	.0175
1.975	.758	.0445	.0175	.0445	.0175	------	------	------	------	------	------	------	------
		Stacking axis coordinates											
		x		y		x		y		x		y	
		cm	in.	cm	in.	cm	in.	cm	in.	cm	in.	cm	in.
		0.810	0.319	0.386	0.152	0.772	0.304	0.361	0.142	0.742	0.292	0.328	0.129

TABLE V. - CHANGE IN EQUIVALENT MASS FLOW WITH AN INCREASE IN RADIAL CLEARANCE

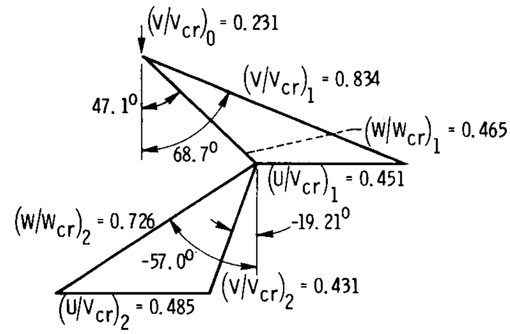
Radial clearance			Mass flow	
Percent	cm	in.	kg/sec	lb/sec
2.3	0.0432	0.017	1.187	2.617
3.3	.0635	.025	1.192	2.628
6.7	.1270	.050	1.198	2.642

TABLE VI. - CHANGE IN EQUIVALENT SPECIFIC WORK WITH AN INCREASE IN RADIAL CLEARANCE

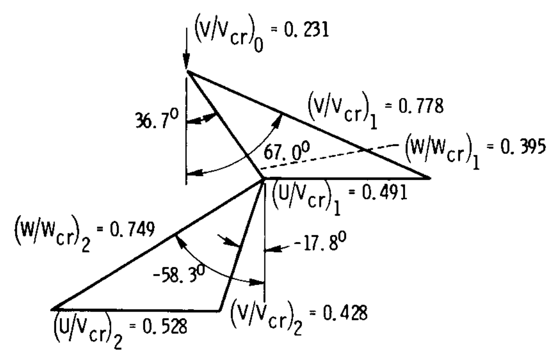
Radial clearance			Equivalent specific work	
Percent	cm	in.	J/g	Btu/lb
2.3	0.0432	0.017	39.572	17.00
3.3	.0635	.025	38.711	16.63
6.7	.1270	.050	36.826	15.82

TABLE VII. - CHANGE IN TOTAL EFFICIENCY WITH AN INCREASE IN RADIAL CLEARANCE

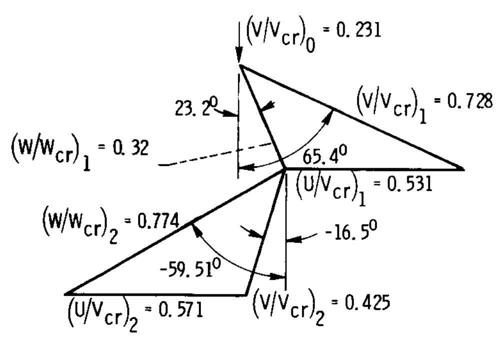
Radial clearance			Total efficiency, percent
Percent	cm	in.	
2.3	0.0432	0.017	88.1
3.3	.0635	.025	86.2
6.7	.1270	.050	82.0



(a) Hub section; radius ratio $r_h/r_t = 0.850$.



(b) Mean section; radius ratio $r_m/r_t = 0.925$.



(c) Tip section; radius ratio $r_t/r_t = 1.000$.

Figure 1. - Turbine design velocity diagram for twisted rotor blades.

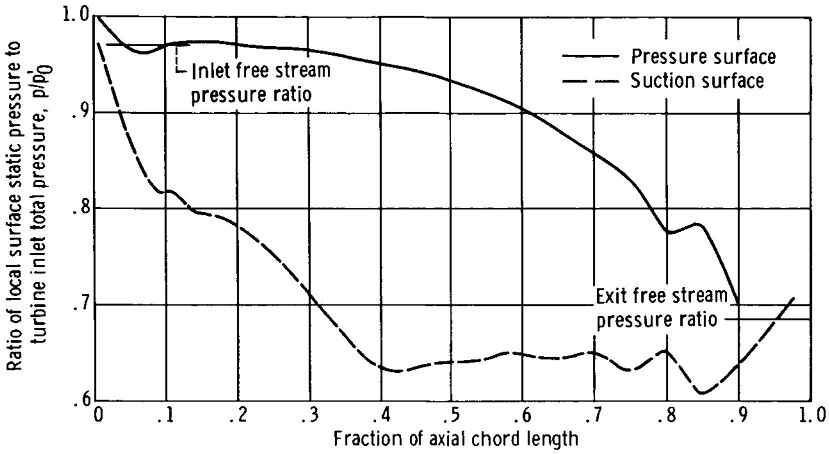


Figure 2. - Design surface static pressure distribution at stator mean section.

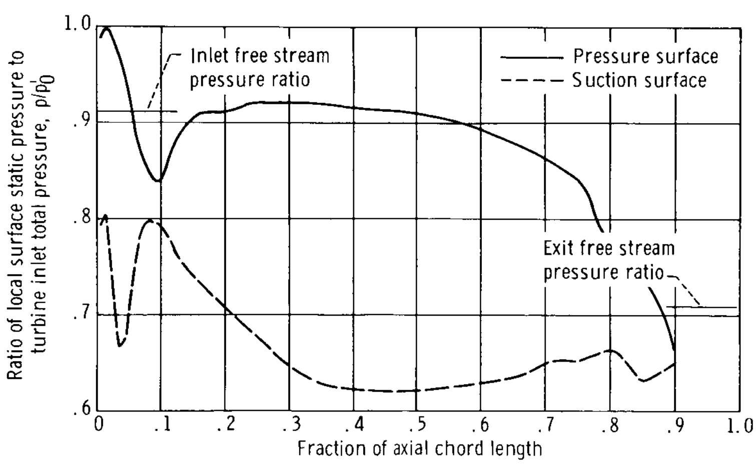


Figure 3. - Design surface static pressure distribution at rotor mean section.

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Figure 4. - Turbine stator.

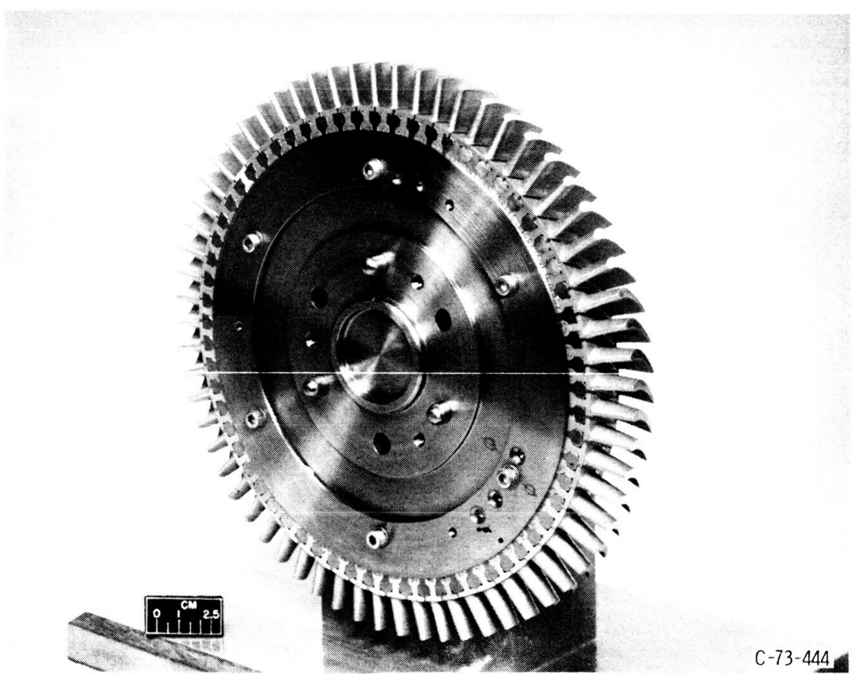

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Figure 5. - Turbine rotor.

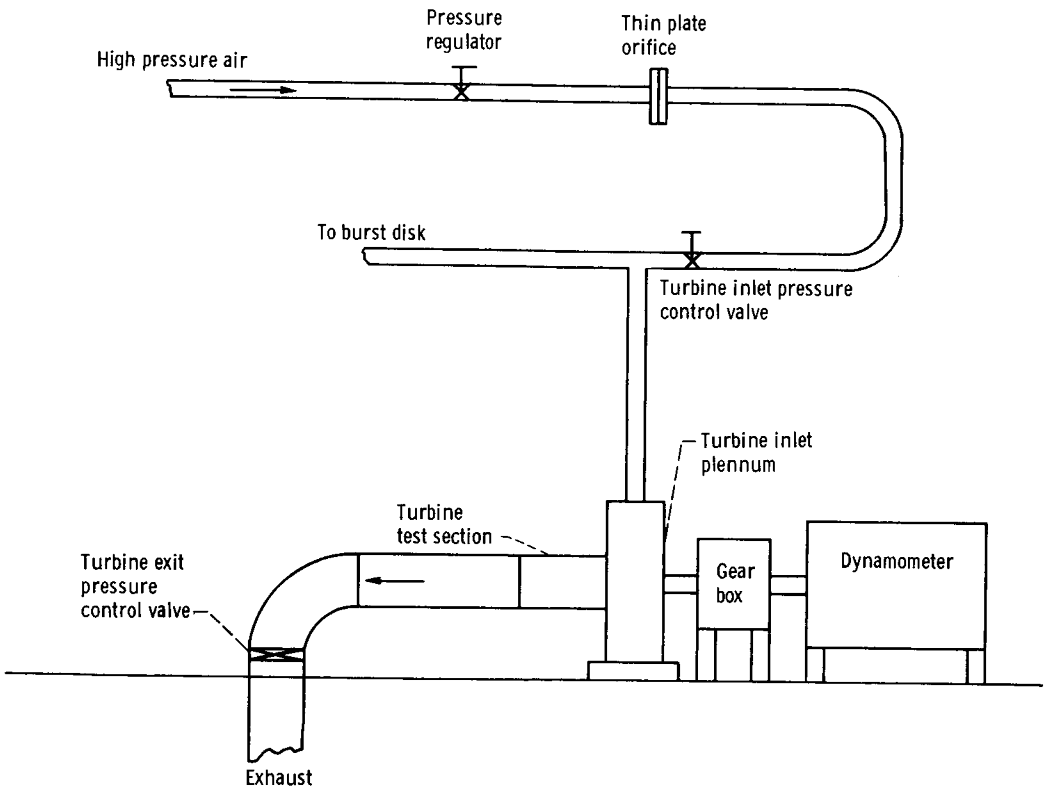

Figure 6. - Test installation schematic.

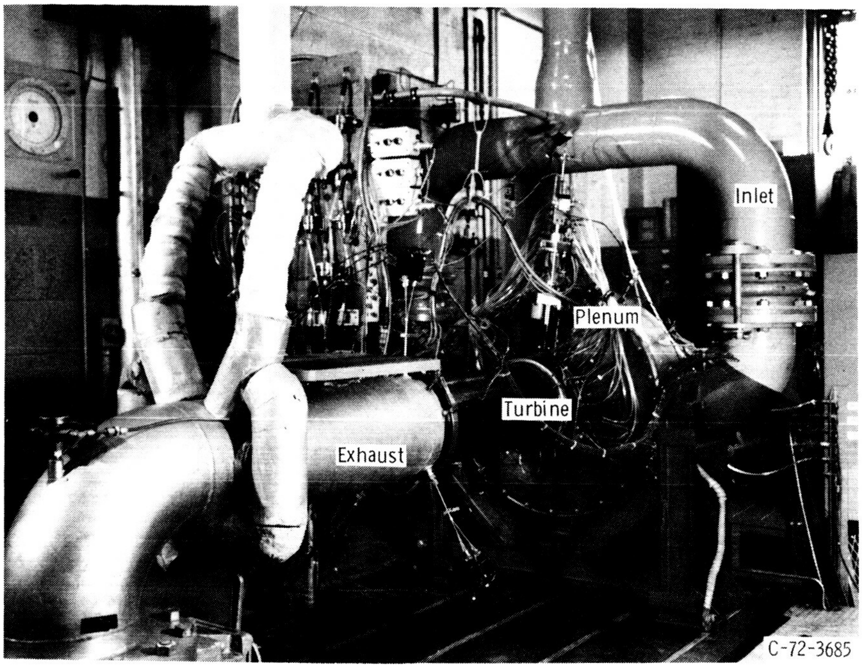

Figure 7. - Test facility.

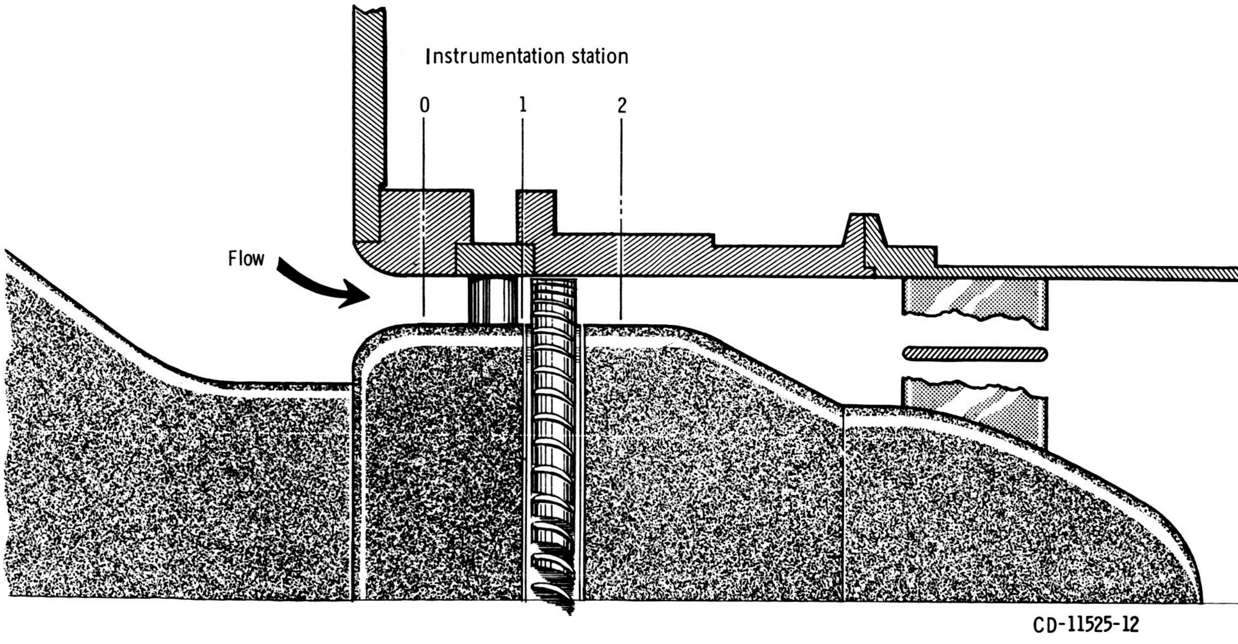

Figure 8. - Schematic of turbine test section.

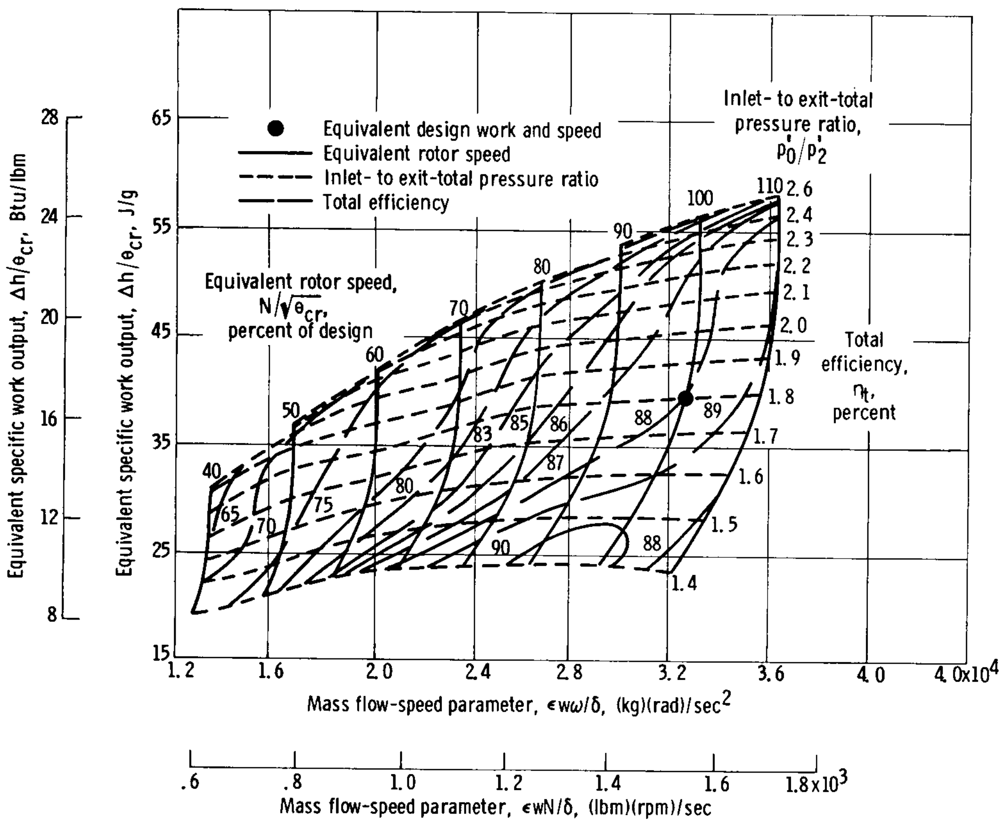

Figure 9. - Overall turbine performance map.

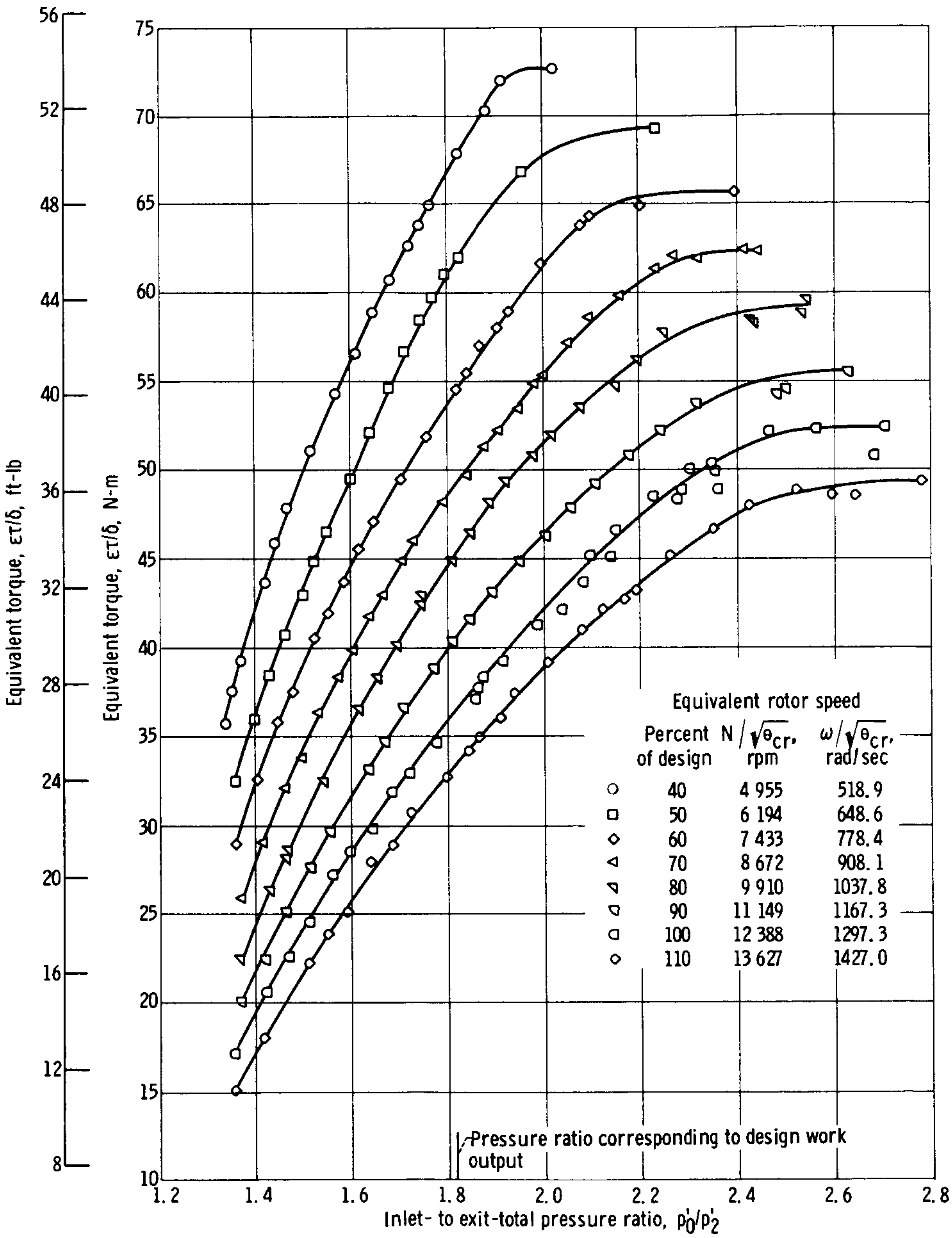

Figure 10. - Variation of torque with pressure ratio and speed for twisted rotor blade configuration.

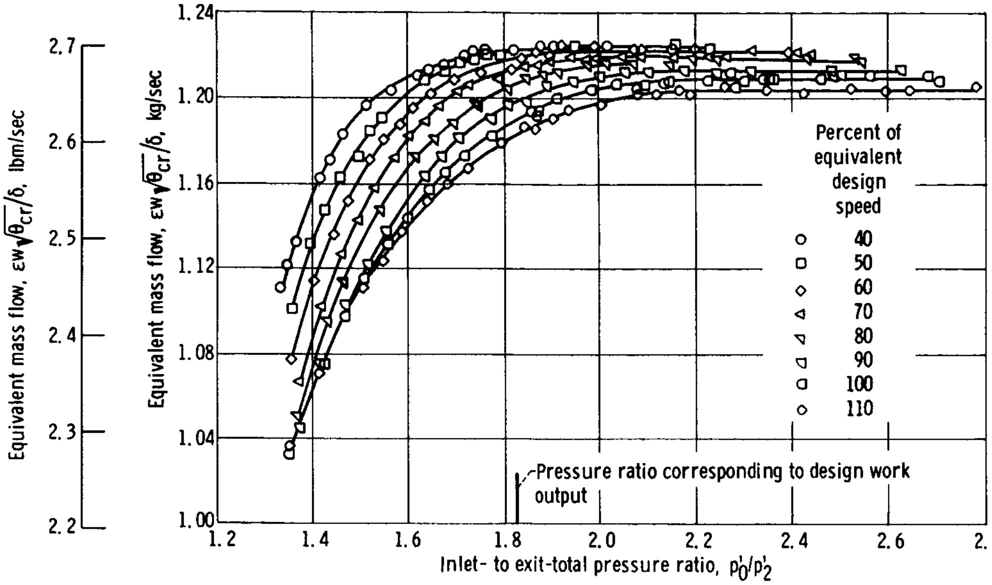

Figure 11. - Variation of mass flow with pressure ratio and speed for twisted rotor blade configuration.

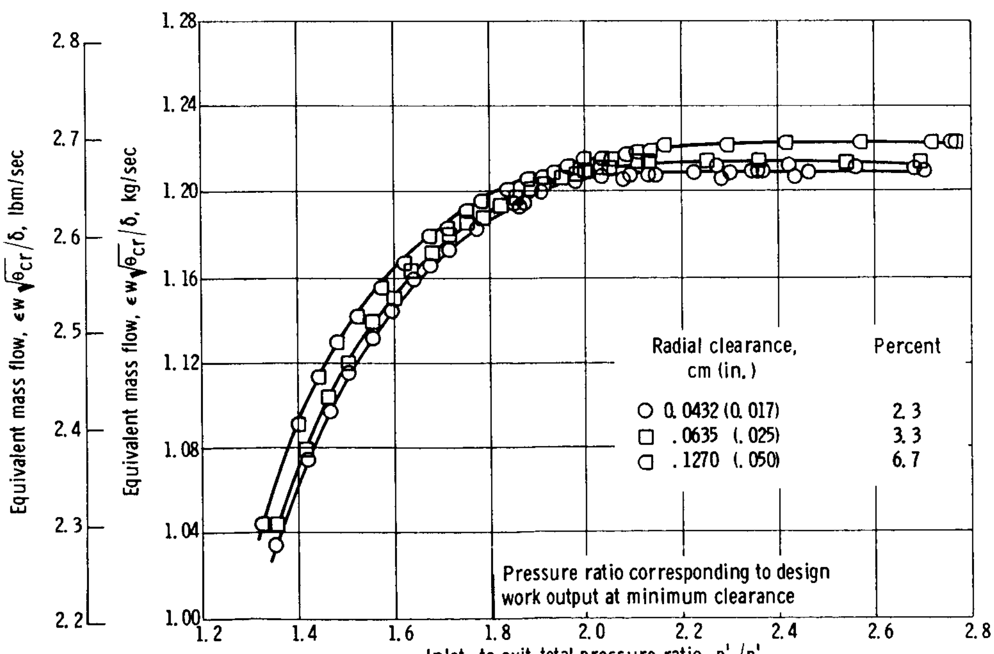

Figure 12. - Variation of equivalent mass flow with pressure ratio at design equivalent speed.

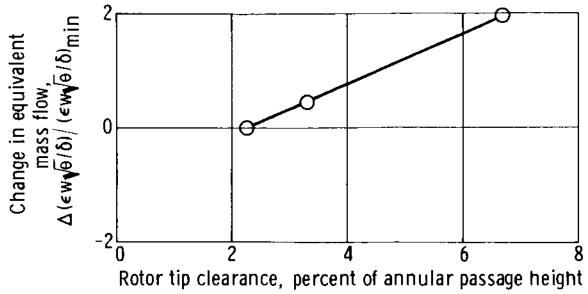

Figure 13. - Change in equivalent mass flow with radial clearance.

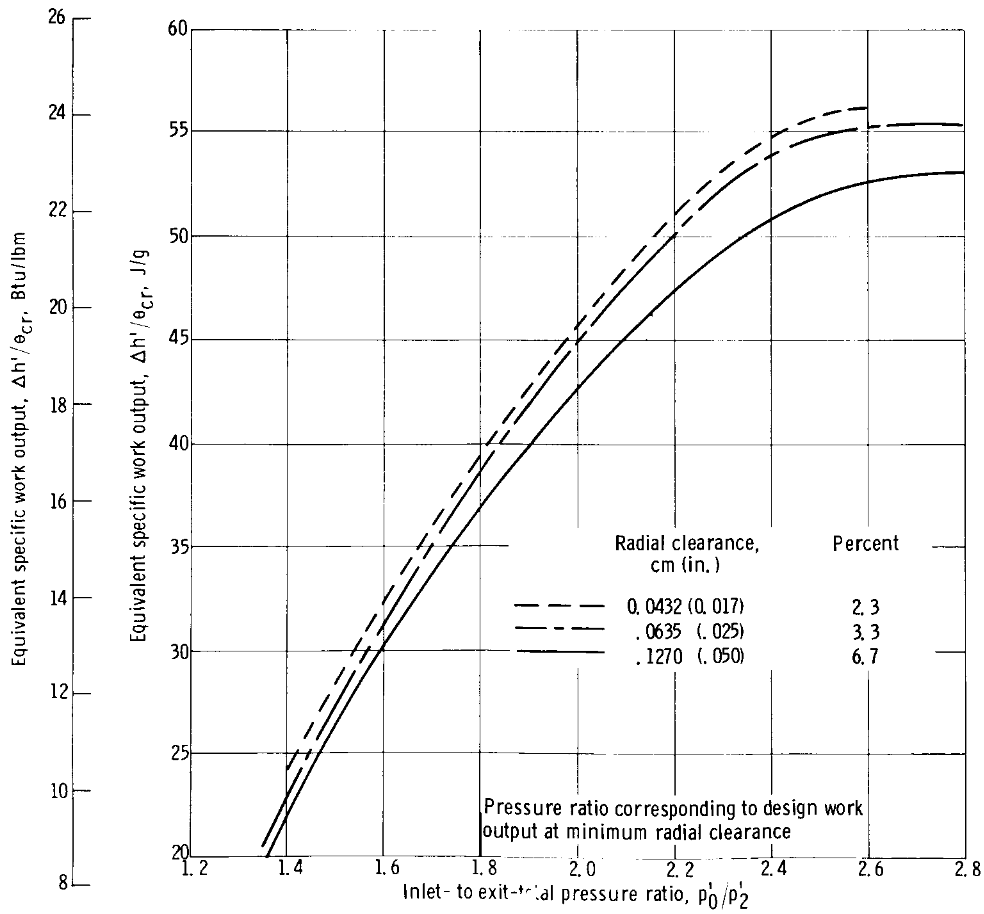

Figure 14. - Variation of specific work output with pressure ratio at design equivalent speed.

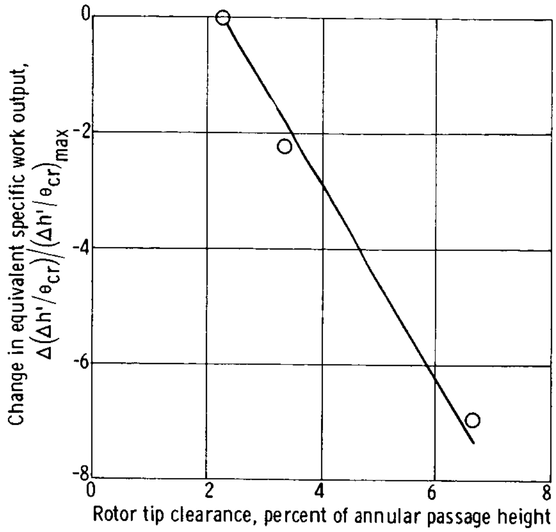

Figure 15. - Change in equivalent specific work output with radial clearance.

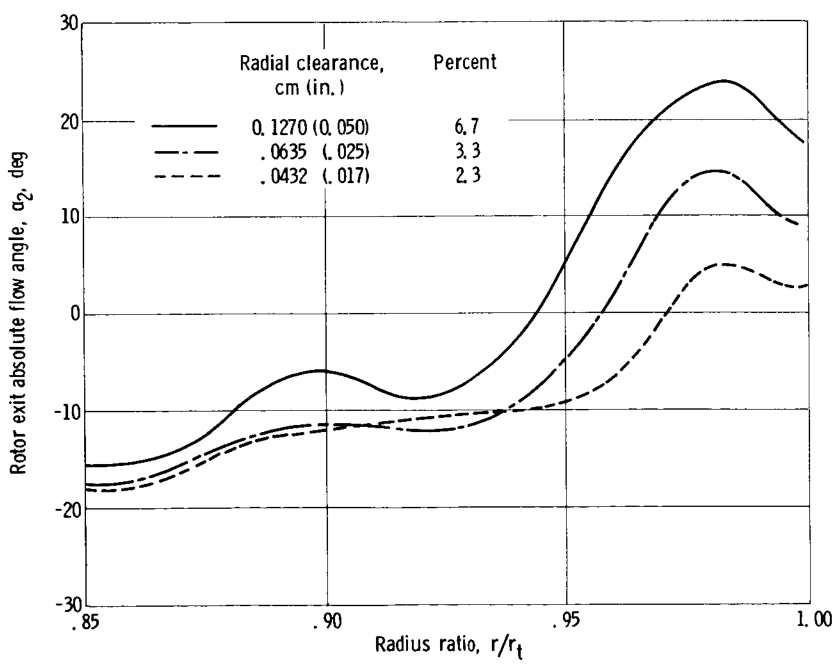

Figure 16. - Variation of rotor exit flow angle with radius ratio at equivalent design speed and inlet- to exit-total pressure ratio of 1.803.

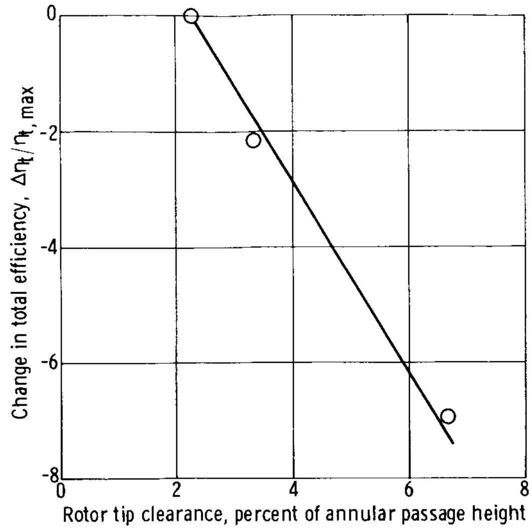

Figure 17. - Change in total efficiency with radial clearance.

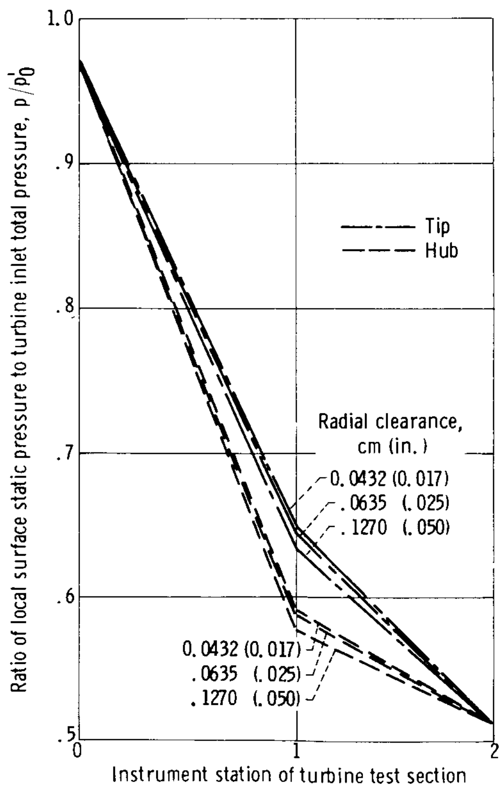

Figure 18. - Comparison of static pressure variation through turbine for various radial clearances at overall inlet- to exit-total pressure ratio of 1.803.

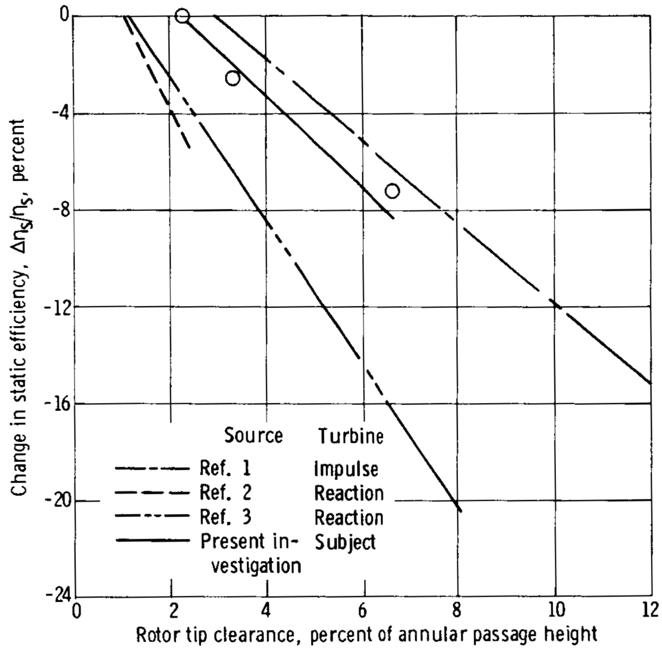

Figure 19. - Effect of rotor tip clearance on performance for various turbines.

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—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

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